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Topic B4: Ventilation

COMPARISON OF RADIANT AND CONVECTIVE COOLING OF OFFICE ROOM: EFFECT OF WORKSTATION LAYOUT

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SUMMARY

The impact of heat source location (room layout) on the thermal environment generated in a double office room with four cooling ventilation systems - overhead ventilation, chilled ceiling with overhead ventilation, active chilled beam and active chilled beam with radiant panels was measured and compared. The room was furnished with two workstations, two laptops and two thermal manikins resembling occupants. Two heat load levels, design (65 W/m^2) and usual (39 W/m^2), were generated by adding heat from warm panels simulating solar radiation. Two set-ups were studied: occupants sitting by the windows, and near the opposite wall. The room air temperature of 26 ^{o}C was kept constant. Air temperature, globe (operative) temperature, manikin based equivalent temperature and air velocities were all measured. All systems performed equally well and managed to keep the required thermal environment.

INTRODUCTION

The present conventional ventilation and air conditioning systems, based on mixing air distribution only, are not able to provide high quality of indoor environment in energy efficient way; often the resulting indoor climate is mediocre and poor (draught discomfort, poor air quality, local temperature asymmetry, etc.), and does not fulfil the requirements set in existing standards and guidelines, EN 15251 (2007), ISO 7730 (2005). To achieve comfortable indoor environment new ventilation and air conditioning strategies are needed. Combining convective and radiant cooling can lead to improved performance of indoor climate systems. If the radiant cooling is provided within the occupied zone, the background (air) temperature can be kept higher: the globe (operative) temperature will be lower. Radiant cooling indoors is provided by using water as working media. Since density of air is 1.12 and of water 1000 kg/m^3 and specific heat capacity of water is 4.18 and of air 1.01 kJ/kg K, i.e. water is 4000 times better heat conductor than air for the same volume of fluid. As a result, the necessary supply air is reduced because it is needed for fresh air supply to fulfill the air quality requirements (EN 15251, 2007). Therefore the duct dimensions, fans and HVAC requirements are reduced which leads to further energy savings (less air is moved and conditioned).

An important factor that has to be considered when applying a ventilation system for particular space is the room lay-out. It defines the heat sources location in the room. Thermal environment and air quality in rooms depend strongly on the interaction of buoyancy, ventilation and other flows. Radiant thermal asymmetry is another factor having impact on occupants' thermal comfort. Sitting close to the window or away during summer requires different airflow distribution in order to ensure comfortable environment.

The performance of both conventional ventilation systems (mixing ventilation) and modern ventilation systems based on pure convective or combined convective and radiant cooling (chilled beam, chilled ceiling with mixing ventilation and chilled beam with radiant panels) under different levels of heat loads have been studied separately but not compared in details, Mustakallio et al. (2013). The performance of these systems under varying heat source strength and location with respect to indoor thermal environment at Category II (EN 15251, 2007) is reported in this paper.

METHODOLOGIES

A full-scale test room, 4.12 m x 4.21 m x 2.89 m (L x W x H), was furnished with two desks. On each workstation there was a laptop to simulate the heat load from office equipment. Two thermal manikins with realistic female body geometry were used to mimic two occupants seated at the desks (Figure 1). The manikins were dressed with typical summer clothes – light shoes, cotton socks, panties, light cotton trousers and a T-shirt. The overall insulation of clothes was 0.5 Clo for each manikin. The manikins had short-hair wigs. Four low energy lamps, 40 W each, were used to provide ambient light in the chamber. Five water radiant panels (total area of 6.3 m^2) with controlled surface temperature were installed to mimic windows heated by direct sunlight. Five heated electrical foils (total area of 8 m^2) were placed along the floor below the windows to generate heat and to simulate direct solar radiation.

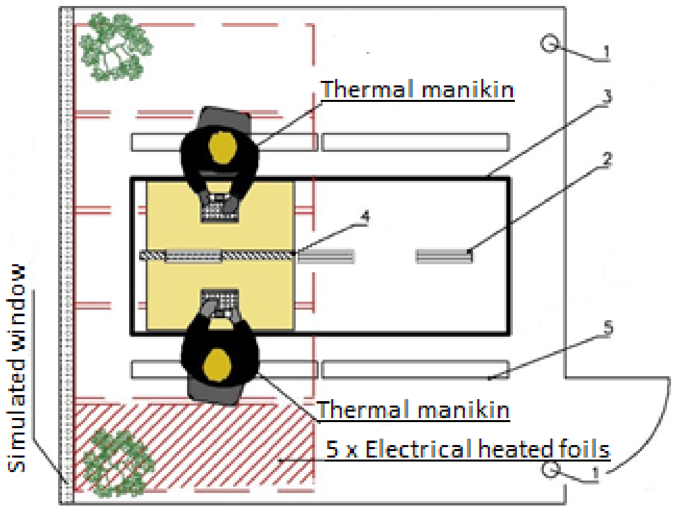

Figure 1. Experimental set-up: 1) exhaust air terminal device; 2) supply air terminal device; 3) chilled beam (CB); 4) desk partition; 5) light fixtures. The electrical heated foils on the floor (red hatched squares) mimicked the solar heat gains from floor.

The test room was equipped with HVAC system able to control the air temperature, the relative humidity and the flow rate of the supplied conditioned air. The performance of four systems to provide comfortable thermal environment within the range of Category II (EN 15251, 2007) based on pure convective or combined convective and radiant cooling was investigated in this study: mixing ventilation alone (MV), chilled beam (CB), chilled beam with radiant panels (CBR) and chilled ceiling combined with mixing ventilation (CCMV).

Mixing ventilation

The mixing ventilation, when used alone, consisted of three linear double-slot diffusers installed in the middle of the suspended ceiling (Figure 1). The supplied air was guided over the ceiling surface via louvers incorporated in the linear diffusers. The air was exhausted by two circular exhaust diffusers mounted in the ceiling, in the two corners opposite to the wall with the simulated windows (Figure 1). The same exhausts were used for all four systems tested.

Chilled Beam/ Chilled Beam with Radiant Panels

The chilled beam (CB) tested in this study consisted of two main components: active chilled beam, with two heat exchangers alongside the plenum box, and a circuit of five hydronic radiant panels integrated in chilled beam design (Halton Oy). The surface area of the radiant panels incorporated in the chilled beam was 3.6 m^2.A 3-way manually operated valve allowed to cut off the water flow towards the radiant panels and bypass them. When the water was flowing into the radiant panels the system used was combining convective and radiant cooling and was referred as chilled beam with radiant panels (CBR). The primary air, supplied from the plenum box, mixed with the entrained air and was discharged upward to flow along the ceiling before entering the occupied zone of the room, i.e. used chilled beams were of "exposed" type. The ratio fresh to entrained air was 1:5.

Chilled Ceiling

The radiant ceiling consisted of 18 panels (Uponor, Comfort panels) with dimensions 1.2 m x 0.6 m. The panels were connected in 6 rows of 3 panels connected in series. The middle row of the ceiling was made of standard ceiling panels in which two air supply diffusers were installed. The ceiling area covered by the radiant panels was 12.64 m^2, which was 77% of the total ceiling area of 16.48 m^2. The air from the diffusers was discharged tangentially along the ceiling, i.e. to be in contact with the cold surface of the ceiling.

Experimental Conditions

Summer conditions under a design room air temperature of 26 ^oC were tested with the four climate systems complying with the upper limit for Category II EN 15251 (2007). Two different heat loads were generated in the experimental room – design heat load or high heat load of 65 W/m^2 (H as "high") and usual heat load or low heat load of 39 W/m^2 (L as "low"), Table 1.

Each system was tested for 2 different room furniture set-ups (lay-outs), Figure 2. The red wall is the simulation window, the blue squares represent the tables and the blue circles are the two occupants (thermal manikins). The empty blue rectangular defines the position of the CB/CBR when it was installed. In both set-ups the occupants were facing each other. The tables were grouped together with a partition in between (Figure 2). The occupants were not sitting exactly below the chilled beam and hence were not directly exposed to the radiant panels build in it. In Figure 2, case S1, the thermal manikins were positioned 0.6 m from the simulated window, in the middle of the simulated solar irradiation on the floor (Figure 1). In S2 the thermal manikins were seated 0.6 m from the opposite wall.

Table 1. Simulated heat load in the test office room

Heat Gain	Design (High) Heat Load [W]	Usual (Low) Heat Load [W]	Amount [-]
Window surface load, [W]	403	201	-
Solar radiation on floor, [W]	250	-	-
Laptop, [W]	65	65	2
Occupants, [W]	87	87	2
Lighting, [W]	40	40	4
Total, [W]	**1117 or 65 W/m^2**	**665 or 39 W/m^2**	-

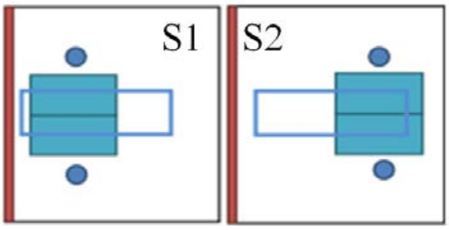

Figure 2: The two room set-ups studied.

Under all cases the supplied air temperature was kept at 16 ^{o}C and flow rate at 26 L/s. In the cases with MV alone the supplied amount of outdoor air was increased to 55 L/s under the usual (low) heat load and to 90 L/s under the design (high) heat load in order to remove the heat load and to achieve air temperature of 26 ^{o}C. When the CCMV and MV cooling systems were tested the chilled beam was dismounted from the ceiling and removed from the room. When the MV system was operated the cold water circulating in the ceiling radiant panels was completely stopped. Hence, as already pointed the whole cooling power provided was from the air.

Measured parameters

Air temperature, globe (operative) temperature, velocity and turbulent intensity were measured and draft rate levels calculated at 8 different heights in the room: 0.05, 0.1 0.3, 0.6, 1.1, 1.7, 2.0 and 2.4 m. For the purpose all sensors were mounted on a stand 2.7 m tall.
Measurement pole locations are shown in Figure 3. The measurements were not performed at locations 1, 6, 11, 16 and 21, because of low air velocity and temperature closed to 28°C was identified in a previous study with three of the systems, i.e. CB, CBR and CCMV, Mustakallio et al. (2013). Because of inaccessibility measurements were not performed at the following points: 7, 12 and 17 for set-up 1, and 9, 14 and 19 for set-up 2;

Surface temperatures and radiant temperature asymmetry were measured as well. They are not reported in this paper. The two thermal manikins were used to measure the local thermal conditions at the occupied zone for the two set-ups, S1 and S2. The results from the thermal manikin measurements are not reported in this paper.

All temperature sensors were of a thermistor type with accuracy of ±0.2 °C. Air temperature was measured with radiation shielded sensors. Velocity sensors were omnidirectional hot-wire anemometers with accuracy of ±0.2 m/s or ±1% of the reading within the range 0.05-1.0 m/s. Prior to the experiment all sensors were calibrated.

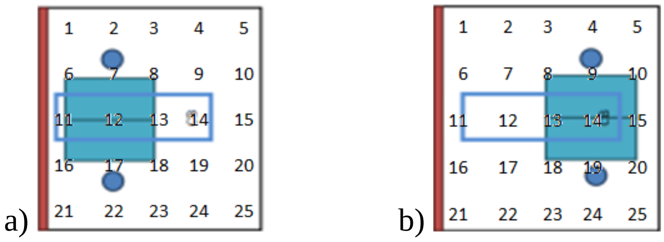

Figure 3. Measuring points locations in the chamber under a) set-up 1 and b) set-up 2.

Experimental Procedure

Before any measurements were started it was ensured that the air temperature measured at the reference point 1 (between measuring points 2 and 3) for room set-up 2, and at reference point 2 (between measuring points 3 and 4) for room set-up 1 was at 26 °C. The purpose was to use as reference that air temperature which was less affected from the heated manikin. The criterion for steady state was stable air temperature at both reference points. It was also checked whether the surface temperature of the thermal manikins was kept constant and within ± 0.1 K change for a 10 min period. If all the conditions were fulfilled the physical measurements were started. Measurements of air temperature, globe (operative) temperature and air velocity at 8 different heights were performed at one measuring point at a time. Logging time was 5 minutes. Then the stand was moved to the next measuring point. After waiting for 5 min the next measurement was initiated. This was needed in order to make sure that the airflow pattern in the chamber was not affected by the movement of the stand. After all the points in the chamber were measured for the tested condition, the manikins were logged for 5 min to obtain their surface temperature and the heat supplied to them. The radiant asymmetry was also measured at the locations where the manikins were placed at the end of each case. Measurements were performed at heights 0.6 m and 1.1 m and in 3 directions.

Data Analysis

Because of the small differences in reference point temperatures among the cases (up to ± 0.4 K), a normalized temperature was used to compare the results. The normalized temperature, TF_a and TF_o, are dimensionless parameters, which are calculated as the average air or globe temperatures, respectively from all the measured points at a particular height divided by the mean reference point temperature for that particular case. It is a ratio which shows how uniform environment the particular system is able to provide in the room – the closer to 1, the more stable the system is.

RESULTS AND DISCUSSION

Air temperature

All systems performed equally well under the two studied set-ups, Figure 4. It is noticed that only between the heights 1.7 m and 2.4 m the air temperature is higher than the reference point temperature of 26 ^{o}C. The reason was that the 4 lightings were hanging 1.7 m from the floor which makes the average air temperature at that height higher compared to the other measured heights. For CCMV and under the two set-ups at both design and usual heat load the TF_a was closest to 1 compared to the other 3 systems tested. With CB at design heat load for set-up 2 the averaged air temperature in the room was lower than in the reference point at all heights (TF_a less than 1). Visualizations performed showed that the air supplied by the CB dropped down faster compared to CBR, CCMV and MV, i.e. the air “dropped” down before reaching the wall where the reference point was set.

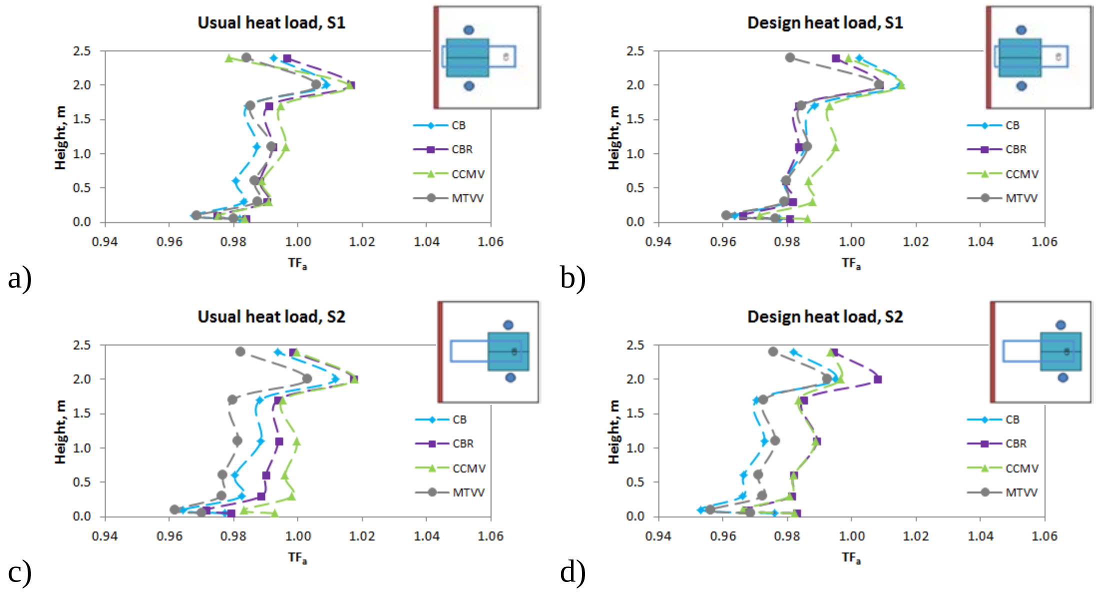

Figure 3. Averaged air temperature factor (normalized temperature) in height for a) set-up 1 usual(low) heat load, b) set-up 1 design(high) heat load, c) set-up 2 usual(low) heat load and d) set-up 2 design(high) heat load.

Globe (operative) temperature

Under the usual (low) heat load the four systems performed equally with respect to the averaged globe (operative) temperature in the space, Figure 4. CCMV provided lower globe (operative) temperature throughout the room than at the reference point at design (high) heat load for both set-ups. The operative temperature decreased with height: the higher from the floor (i.e. closer to the panels) the more significant the cooling effect was. In these results the average globe temperature was calculated from all measuring points in the grid for that height, TF_o, and since the chilled ceiling was across the whole ceiling, the cooling effect can be clearly seen. However, this tendency was present with the design/high heat load only. With the usual (low) heat load the globe temperature was kept stable throughout the room by all systems. The supply temperature of the water and its flow were also varied with the heat load in the room for all systems of combined convective and radiant cooling: lower flow and higher supply temperature of the water were set for the usual (low) heat load conditions, Table 2. A decrease in the globe temperature with height was not observed with CBR for either of the studied heat loads or set-ups. The radiant cooling from CBR had a local effect on several measuring points only. The impact was not great enough to affect the averaged value.

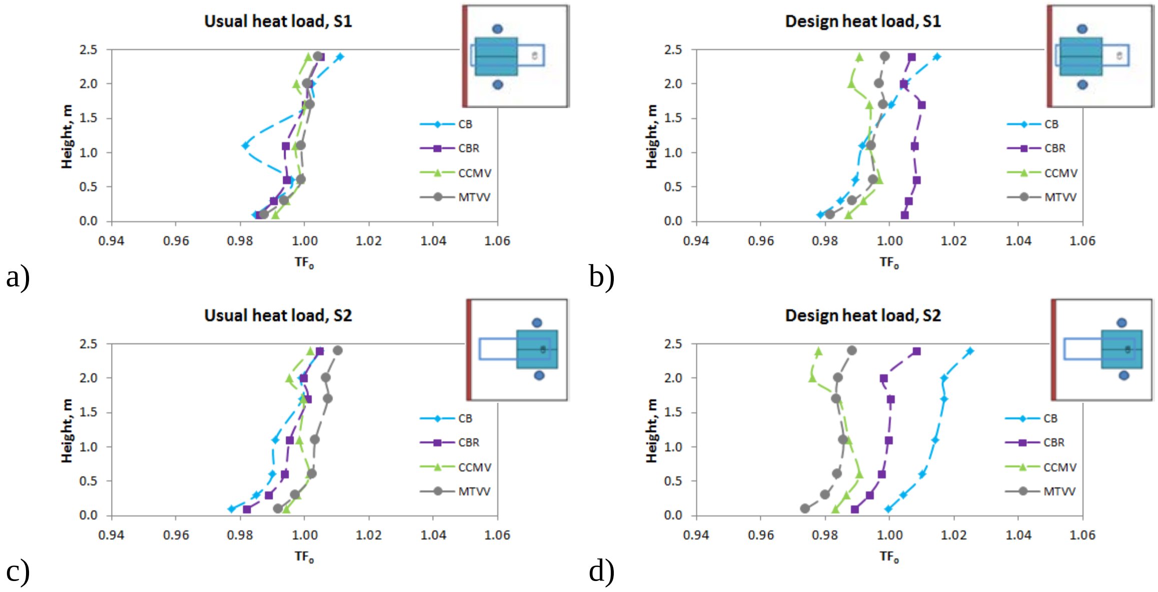

Figure 4. Averaged globe (operative) temperature factor (normalized globe (operative) temperature) in height for a) set-up 1 usual(low) heat load, b) set-up 1 design(high) heat load, c) set-up 2 usual(low) heat load and d) set-up 2 design(high) heat load.

Air velocity

With all four systems the highest averaged air velocities were measured close to the floor between 0.05 m and 0.1 m, Figure 5. With all four systems the averaged air velocities were below the recommended limit of 0.2 m/s (ISO 7730, 2005). The air supply jets reached the walls, fell down and then glided along the floor. Velocities were also slightly higher at 2 m. This is because of the supply air jets spread. Air velocities in the occupied zone were lower than 0.2 m/s, ISO 7730 (2005).

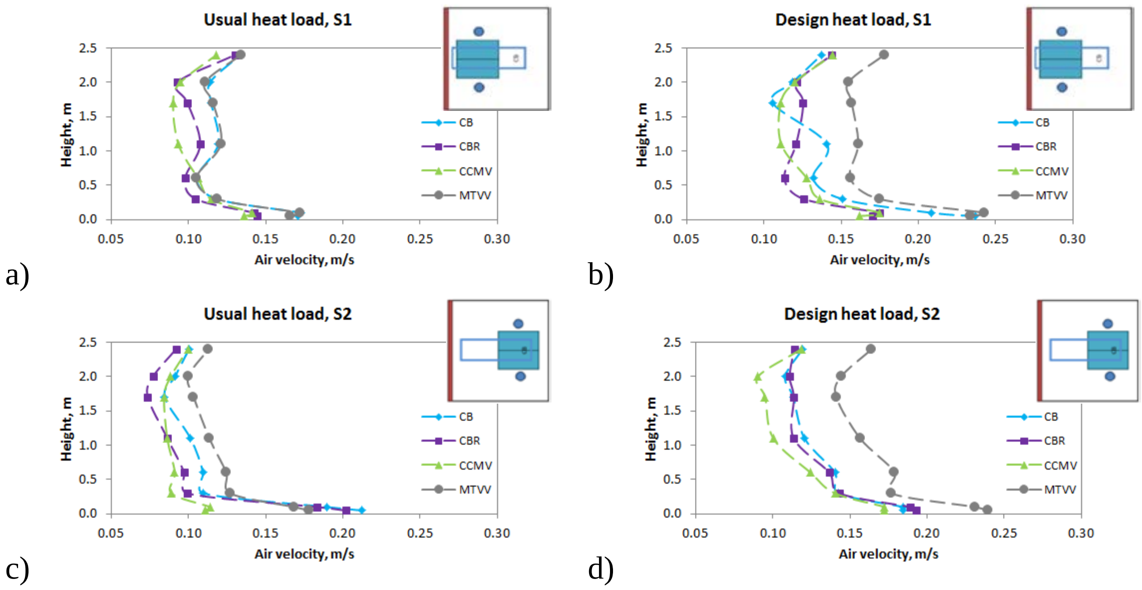

Figure 5.Averaged air velocity in height for a) set-up 1usual(low) heat load, b) set-up 1 design(high) heat load, c) set-up 2 usual(low) heat load and d) set-up 2 design(high) heat load.

Slightly lower velocities were achieved with CCMV. Although the air supply flow rate is the same as for the CB and CBR, i.e. 26 L/s, air velocities were lower with CCMV at most of the cases. The total amount of air with the CB and CBR was in fact 130 L/s: primary and entrained room air flow leading to higher velocities with these two systems. The highest velocities were achieved with mixing ventilation (MTVV), which was expected because mixing ventilation cooling power comes from the supplied conditioned air in the room. In order to remove the heat load, relatively high supply air flow rate was needed (55 L/s under usual/low heat load and 90 L/s under high heat load). For the remaining three systems only 26 L/s were needed for both heat loads. The rest of the cooling load was covered by radiation, i.e. the water. Room set-up had a little impact on the air velocity for all four systems resulting in similar velocity profiles, Figure 5.

CONCLUSIONS

In this study the thermal environment generated by four indoor climate systems based on pure convective or combined convective and radiant cooling were investigated: chilled beam, chilled beam with radiant panels, chilled ceiling with mixing ventilation and overhead mixing ventilation alone. The following conclusions can be made:

- All systems managed to keep the required thermal environment for Category II EN 15251 (2007).
- Heat load and room set-up had insignificant effect on the systems` performance with regard to averaged air temperature in the space.
- During design (high) heat load conditions chilled ceiling combined with mixing ventilation was able to provide lower averaged globe (operative) temperature in the room than the other three systems.
- During usual (low) heat load conditions all systems provided equally homogenous environment with regard to globe (operative) temperature.
- The highest air velocities were near the floor and decreased with height for all studied systems.
- Lowest air velocities were provided by chilled ceiling combined with mixing ventilation (CCMV).
- Highest air velocities are measured with mixing ventilation (MV).
- Room set-up does not have significant impact on air velocities in the room.

ACKNOWLEDGEMENT

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